N66 28764
(ACCESSION NUMBER) (THRU)
58 1
(PAGES) (CODE)
CR-75768 15
(NASA CR OR TMX OR AD NUMBER) (CATEGORY)

FACILITY FORM 602

GPO PRICE $ ______

CFSTI PRICE(S) $ ______

Hard copy (HC) 3.00

Microfiche (MF) .50

ff 653 July 65

JET PROPULSION LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
PASADENA, CALIFORNIA

RE-ORDER NO. 65-1099

ASTRIONICS DIVISION

DESIGN, FABRICATION, MODIFICATION, AND TESTING

of

CRYOGENIC-SOLID COOLER

AEROJET MODEL K6

Final Engineering Report

to

Jet Propulsion Laboratory

Pasadena, California

Contract No. 950726

Report No. 2948 / 15 Nov. 1965 / Copy No.

This work was performed for the Jet Propulsion Laboratory, California Institute of Technology, sponsored by the National Aeronautics and Space Administration under Contract NAS7-100.

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NOTE: The information contained herein is to be regarded as preliminary and subject to further checking, verification, and analysis.

RE-ORDER No. 65-1099

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ABSTRACT

The design, development, and fabrication of Cryogenic-Solid Cooler Aerojet Model K6, as well as subsequent testing and experimentation, are reported herein.

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Report No. 2948

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APPENDIX A - Instructions for Operating Cryogenic-Solid Cooler Model K6

I. INTRODUCTION

The purpose of this program was to design, fabricate, test, and deliver a solid cryogenic-system suitable for use in performing infrared detector cooling experiments. A study was performed initially to establish the design criteria for a system capable of cooling a 0.1-watt detector heat load to 77°K for 1 year.

Results of subsequent tests with the fabricated cooler showed that approximately six months of life and a temperature of 80° to 85°K would be commensurable with the size of the cooler and the heat loading properties of the materials used.

The resultant cryogenic-solid cooler, designated Aerojet Model K6 (Figure 1), consists of a coolant container, fill port, vent port, heat transfer rod between the detector and the coolant, temperature control system, and temperature sensing devices. A flexible detector mount and safety valve were modifications effected after completion of the system as was the portable cooler cart (Figure 2).

The cryogenic-solid cooling system has attained temperatures of 79°K and 65°K using solidified methane and nitrogen respectively. These temperatures were held within ± 0.5°K for extended periods. Through the use of experimental data, the operating life of the system was determined to be 6.9 months with a total system heat load of .535 watts and solid methane as the coolant.

The standby characteristics of the system are quite excellent. When using solid N_2 as a coolant, the temperature of the heat transfer rod merely increased from 50°K to 51°K over an 8-hr period with the pumping valve closed.

II. THEORETICAL CONSIDERATIONS

A. HEAT-LOAD CALCULATIONS AND CORRESPONDING METHANE REQUIREMENTS

It was intended that the cryogenic-solid cooler would operate with solidified methane (Figure 3) at a temperature of $77^{\circ}K \pm 0.5^{\circ}K$ for 1 year without refilling. Accordingly, the heat load to be contributed by each discrete area was calculated, except for that of the detector which was stipulated as 0.1 watt in the contract. The amount of methane required for dissipating the total heat load was then calculated as the sum of the following individual loads.

1. Methane Required for Fixed Detector-Heat-Load

The amount of Methane required for 1-year operation considering only the 0.1 watt detector heat load was calculated as

$$W_c = \frac{(Q)(t)(C)}{H_s} \qquad (1)$$

where W_c = weight of coolant, lb.

Q = heat load, watts.

t = time in hours (1 year = 8760 hr).

H_s = heat of sublimation, cal/lb.

C = a constant, 860.4 cal/hr.

Therefore, the volume of methane in lb (H_s = 61,644 cal/lb) is

$$W_c = \frac{(0.1)(8760)(860.4)}{61,644} = 12.23 \text{ lb}$$

2. Methane Required for Detector Mount

The heat load contributed by the detector mounting assembly, as determined by its design, was not expected to exceed 0.05 watt. The amount of methane required for this load would be:

$$W_c = (122.3)(0.05) = 6.12 \text{ lb}$$

3. Methane Required for Coolant Container Supports

The heat load introduced by the coolant container supports was based on the anticipated use of Dacron cords having an 8×10^{-5} sq. in. cross-section (0.01 in. dia) and being 2 in. long:

$$Q = \frac{k \; A \, \Delta \, T}{\;} \qquad (2)$$

where Q = heat load, watts

k = thermal conductivity

A = cross sectional area

ℓ = length of cords, in.

ΔT = temperature difference between ends of cords.

For Dacron, $k = 1.2 \times 10^{-3} \dfrac{\text{watts-in.}}{\text{in}^2 \, {}^{o}\text{K}}$, and in this application,

$\Delta T = 223^{o}K.$

Thus, $Q = \dfrac{(1.2 \times 10^{-3})(8 \times 10^{-5})(223)}{2} = 1.07 \times 10^{-5}$ watts

If there were to be 24 support cords, the total heat load would be $(24)(1.07 \times 10^{-5}) = 2.6 \times 10^{-4}$ watts.

Thus, the quantity of methane calculated for this heat load is:

$W_c = (122.3)(2.6 \times 10^{-4})^{2} = 0.0318$ lb

4. Methane Required for Vent Port

It was assumed that the vent port would consist of a 0.75-in. dia and 40-in long coiled tube of stainless steel with a wall thickness of 0.010 in. Again, heat flow Equation (2) is applied after the cross sectional

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area is determined as:

$$A = 0.7854\ (D_o^{\ 2} - D_i^{\ 2}) = (0.7854)(0.77^2 - 0.75^2) = 2.4 \times 10^{-2}\ \text{in}^2$$

Since $$k = 0.203\ \frac{\text{watt-in.}}{\text{in}^2\ {}^oK}$$

Thus, $$Q = \frac{(0.204(2.4 \times 10^{-2})\ (223)}{40} = 2.72 \times 10^{-2}\ \text{watts}$$

$$W_c = (122.3)(2.72 \times 10^{-2}) = 3.33\ \text{lb}$$

5. Methane Required for Fill Port

The fill port does not have to be as large as the vent port since the coolant is pumped into the container under pressure. Therefore, assuming a 0.25-in. dia stainless steel fill-port having a wall thickness of 0.006 in. and a length of 40 in., the same equations used for the vent port apply:

$$A = (0.7854)(0.262^2 - 0.25^2) = 6.9 \times 10^{-3}\ \text{in.}^2$$

Therefore, $$Q = (1.132)(6.9 \times 10^{-3}) = 7.8 \times 10^{-3}\ \text{watts}$$

and $$W_c = (122.3)(7.8 \times 10^{-3}) = 0.954\ \text{lb}$$

6. Methane Required for Insulation Heat Load

Determining the heat leak through the insulation required an assumption of the coolant container size and insulation thickness. Since the outer diameter as well as the external length of the cooler were to be 18 in., a convenient coolant container size was thought to be 14 in. in diameter and length. Based on previous experience and assuming the use of methane, the capacity of a coolant container of this size would be adequate for the present application and would include a compensating factor for detector mount heat load. The heat leak into this 14-in. L = D container is expressed by the equation:

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$$Q = \frac{2\pi k \Delta T}{\alpha} \left\{ \frac{D_c + 2S}{\ln \frac{D_c + 2S}{D_c}} + \frac{D_c^2}{4S} \right\} \qquad (3)$$

where k = the effective thermal conductivity of the insulation

ΔT = the temperature difference

D_c = the diameter of the coolant container

S = the insulation thickness

α = a correction factor.

The value of α can vary from 1 to 6, depending on the insulation technique employed. This value is based on experimental results observed during the development of a number of cooling systems. For the calculation made herein, the value of α is assumed to be 4. The value of k of the Aerojet insulation is $2 \times 10^{-6} \frac{\text{watt-in.}}{\text{in}^2\text{-}^\circ\text{K}}$, D_c is 14 in., S = 1 in., and ΔT is 223°K for the calculation.

Thus, the heat leak is

$$Q = \frac{2 \quad (2 \times 10^{-6})(223)}{4} \quad \frac{16}{\ln \ 1.14} + 49$$

$$= (226)(171)(\pi)(10^{-6}) = 0.120 \text{ watts}$$

Therefore, the amount of coolant required would be:

$$W_c = (122.3)(0.120) = 14.68 \text{ lb}$$

7. Methane Required for Total System

The total amount of methane in pounds for the assumed system is calculated by adding the methane requirement imposed by each component listed as follows:

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	COMPONENT	METHANE REQUIREMENT (lb)
a.	Detector	12.33
b.	Detector Mount	6.12
c.	Supports	0.03
d.	Vent Port	3.33
e.	Fill Port	0.95
f.	Insulation	14.68
	TOTAL	37.44

The 40.76-lb. methane capacity of the 14 in. L = D cylindrical coolant container is well within the capacity calculated (37.44 lb) to affect the total heat load anticipated. This 14-in. dia by 14-in. long coolant container was considered adequate for enclosing within the 18-in. dia by 18-in. high outer container originally envisioned.

B. CALCULATED HEAT LOAD AND OPERATIONAL LIFE WITH OTHER THAN METHANE

Although coolants other than methane can be used with the cooler designed on this contract, the amount would always have to be limited to less than 45 lb by the strength of the coolant support members. Additional or stronger supports which would permit heavier coolant charges would incur corresponding increases in heat loads.

The use of other coolants (solidified nitrogen at 50°K, solidified hydrogen at 12.5°K, and liquid helium at 6°K) would also affect the original heat loads calculated on the basis of the characteristics of methane. This variation would be calculated as the ratio of the difference between the ambient and coolant temperatures, Δ T coolant, of coolants other than methane and the ambient and coolant temperatures, Δ T methane, of methane. The 0.1-watt detector heat load stipulated on this contract would, of course, remain fixed regardless of the coolant used. Using the heat loads calculated for the methane system, appropriate corrections are made in the subsequent sections for the use of solidified nitrogen, solidified hydrogen, solidified neon, and liquid helium.

	COMPONENT	HEAT LOAD (WATTS)
1.	Support	0.1×10^{-3}
2.	Vent Port	27.20×10^{-3}
3.	Fill Port	7.80×10^{-3}
4.	Insulation	120.00×10^{-3}
5.	Detector Mount	50.00×10^{-3}
		205.01×10^{-3}

1. Heat Load and Life With Solidified Nitrogen (65°)

Assuming the use of solidified nitrogen (Figure 4) at a temperature of 65°K, a temperature difference ratio of 235°K for nitrogen to 223°K for methane exists, $\frac{235}{223} = 1.054$, such that the heat load for a nitrogen loaded system would be:

$$Q_N = (205.01 \times 10^{-3})(1.054) = 216.08 \times 10^{-3} \text{ watts}$$

$$Q_{Detector} \qquad 100 \times 10^{-3} \text{ watts}$$

$$Q_{Total} \qquad 316.08 \times 10^{-3} \text{ watts}$$

Based on this heat load calculation, the operational life for a cooler system with a full charge (79 lb) of solidified nitrogen would be

$$t = \frac{W_c H_s}{Q B} \qquad (4)$$

where t = time, days

W_c = weight of coolant, lb

H_s = the heat of sublimation of the coolant, cal/lb or heat of vaporization of liquid helium

Q = the heat load, watts

B = a constant, 20,860 $\frac{\text{cal}}{\text{watt-day}}$.

Thus, $t = \frac{(79)(23,353)}{(0.31608)(20860)} = 279.8$ days or

$$\frac{279.8}{30} \approx 9.3 \text{ months}$$

If only a 40-lb charge of nitrogen were to be used (the weight limit of the coolant container supports) the life would be:

$$t_{40} = \frac{9.3}{79}(40) = 4.7 \text{ months.}$$

2. Heat Load and Life With Solidified Hydrogen (15°K)

The coolant container has a capacity of 6-lb solid hydrogen (Figure 5). The temperature correction factor is $\frac{285}{223} = 1.28$, resulting in a system heat load total of:

$$\begin{aligned} Q_H &= (205.01 \times 10^{-3})(1.28) = 262.41 \times 10^{-3} \text{ watts} \\ Q_{Detector} &= \underline{100 \quad \times 10^{-3} \text{ watts}} \\ Q_{Total} &= 362.41 \times 10^{-3} \text{ watts} \end{aligned}$$

The operating life would be:

$$t = \frac{(6)(55388)}{(0.36241)(20860)} = 44 \text{ days or}$$

$$t = \frac{44}{30} \approx 1.5 \text{ months.}$$

3. Heat Load and Life With Solidified Neon (20°K)

Calculating on the basis of a 40-lb charge of solid neon, the correction factor of $\frac{280}{223} = 1.26$ would apply resulting in a total heat load of:

$$\begin{aligned} Q_{Ne} &= (205.01 \times 10^{-3})(1.26) = 258.3 \times 10^{-3} \text{ watts} \\ Q_{Detector} &= \underline{100 \quad \times 10^{-3} \text{ watts}} \\ Q_{Total} &= 358.3 \times 10^{-3} \text{ watts} \end{aligned}$$

The operating life would be:

$$t = \frac{(40)(11500)}{(0.3583)(20860)} = 61.5 \text{ days}$$

or

$$t = \frac{61.5}{30} = 2 \text{ months.}$$

4. Heat Load and Life With Liquid Helium (6^oK)

To obtain temperatures below 12^oK, it would be necessary to use liquid helium (Figure 7) as a coolant. Helium must be used in liquid form since it is impossible to solidify by reducing the vapor pressure over it. The coolant container has a 2100 cu. in. liquid capacity which represents 9.48 lb. The temperature conversion factor of $\frac{294}{223} = 1.32$, thus,

$$\begin{aligned} Q_H &= (205.01 \times 10^{-3})(1.32) &&= 270.6 \times 10^{-3} \text{ watts} \\ Q_{Detector} & &&= \underline{100 \quad \times 10^{-3} \text{ watts}} \\ Q_{Total} & &&= 370.6 \times 10^{-3} \text{ watts} \end{aligned}$$

and the operating life would be

$$t = \frac{(7)(2220)}{(0.3706)(20860)} = 2 \text{ days.}$$

C. TEMPERATURE CONTROL TECHNIQUES

Originally it was intended that this cryogenic-solid cooling system should attain a temperature of $77^oK \pm 0.5^oK$ is desired. The temperature obtained with a particular coolant depends directly on the pressure maintained over the coolant. Figure 3 is plot of temperature vs vapor pressure over solidified methane. This curve illustrates that in order to achieve 77^oK, the pressure over the solid must be held between 8 to 10 Torr. For a lower temperature, using solid hydrogen as a coolant, the vapor pressure versus temperature curve is shown in Figure 5. In this case, to hold a temperature of 13°K, the pressure over the solid must be between 21 to 39 Torr.

Three temperature control techniques were investigated in an effort to determine the most suitable for the present application. These techniques included (1) a temperature-actuated pressure control valve; (2) a pressure contactor; and (3) a variable orifice. The first method can be designed as part of the cooler and requires no external power. The second method is external to the cooler and requires a small source of power to operate a solenoid valve. The third method is also external but in this case it requires no power. The following sections contain description and theoretical evaluation of each system.

1. Temperature-Actuated Pressure Control Valve

All temperature control techniques entail monitoring and controlling the pressure over the solid coolant. This particular temperature-actuated pressure control valve operates on the principle of a gas thermometer. The device (Figure 8) consists of a temperature-sensing bulb, a bellows chamber, and a relief valve. The bulb is located as close as possible to the detector and measures its temperature. The temperature changes at the detector cause a change in the pressure of helium gas in the sensing bulb. This pressure change is transported through the gas to the bellows chamber causing it to expand or contract, thus opening or closing the relief valve. The relief valve is located in the vent (pumping) port of the cooler. The pressure over the solidified coolant is raised or lowered, thereby causing an increase or decrease in the coolant temperature. Changes in coolant temperature are reflected by changes in the detector temperature; the changes being in the same direction, i.e., lower pressures cause lower temperatures. With this device, the detector temperature is measured and controlled without regard for the equilibrium temperature of the solid coolant.

One disadvantage of this device is that the precision of temperature control depends on the temperature to be controlled. For a given sensing gas pressure, which is governed by the strength and flexibility of the bellows, the most precise control occurs at the lower temperatures.

Over the relatively low pressure range of the sensing gas and the temperature range from $300^{\circ}K$ to $10^{\circ}K$, the helium gas used in the measuring device behaves as a perfect gas. Thus,

$$\frac{P_o V_o}{T_o} = \frac{P_1 V_1}{T_1} \tag{5}$$

assuming the system volume remains constant.

$$P_1 = \frac{P_o T_1}{T_o} \quad \text{or} \quad P_1 = C_o T_1 \tag{6}$$

where $C_o = \dfrac{P_o}{T_o}$

Thus, the magnitude of the pressure changes in the sensing gas varies with temperature, depending on the initial charging conditions. It can be seen that in order to cause a large pressure variation with temperature, required for proper temperature control, a high charging pressure and a low temperature are needed.

The following is an example of typical valve operation using solid methane or solid hydrogen in the system. Assuming a vent port tube having a vent area diameter of 0.307 in. and a valve with a 1-in. dia seat, a fully open valve would have to move h inches, since

$$A = \pi D h. \text{ For A to equal 0.307 in., and} \quad (7)$$

$$h = \frac{A}{\pi D} = \frac{0.307}{0.785} = 0.39 \text{ in.} \quad (8)$$

A typical bellows and spring combination would have a movement of 0.1 in. per lb of pressure change. Such a device would have a maximum allowable pressure of about 40 psi. Therefore, a charging pressure of 30 psi could be used. The bellows sensing chamber is charged with helium gas at the temperature where control is to be maintained for a high C_o value (Equation 6). With methane, this would give a C_o value of $\frac{30}{77} = 0.39$. Therefore, each degree of temperature change near 77°K causes a pressure variation of 0.39 psi. Assuming a bellows area of 1 sq. in., a 10°K temperature change would be required to fully open the valve. This valve would not be particularly sensitive to the $\pm$ 0.5°K temperature changes required at 77°K. At solid hydrogen (15°K) temperatures, however, C_o would be $\frac{30}{15} = 2$, thus, a 1°K temperature change around 15°K would yield a 2-lb pressure change. A valve in this region would have much greater sensitivity to temperature changes thereby providing better temperature control.

2. Pressure Contactor

To overcome the loss of sensitivity limitation passed by the temperature-actuated pressure control concept at higher temperatures, an external control device may be employed. Such a device would consist of a pressure contactor, a relay, and a solenoid valve (Figure 9). The pressure contactor makes or breaks electrical contact in response to pressure changes.

Over a 0-to 20-Torr range, the precision of contacting is 0.1 Torr. The contactor activates a relay which in turn energizes a solenoid valve. The contactor, upon sensing the pressure in the vent line (or inside the coolant container), opens or closes the valve in the pumping line. For higher temperatures (around 77°K with methane) this corresponds to a temperature control well within the required $\pm$ 1 Torr tolerance required for $\pm$ 0.5°K. At the lower temperatures (15°K) even more precise temperature control is possible.

Another advantage of this sytem is the ease with which the pressure controller can be set for pressures corresponding to temperatures other than 77°K. This adjustment of the valve is manual and external to the cooling system.

One drawback of the pressure contactor is that external power is required for operation of the relay and solenoid valve. However, by careful selection of components, this power requirement may be kept at a minimum.

3. Orifice Control

A third method of temperature control that is theoretically feasible, especially at the lower temperatures, involves the orifice principal. Assuming constant detector heat load, system heat load, and rate of evacuation of gas, an orifice of a precise dimension along the vent line will maintain a given pressure over the solid coolant indefinitely. The orifice being in the line of the cooler, will restrict the gas flow.

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The evacuation rate of the sublimed gas is dependent on the pressure difference between the coolant container and the pumping system or evacuated space. For practical purposes, the pressure inside the coolant container will be much greater than the pressure at the pumping system therefore, the pumping system pressure can be considered constant. Consequently, it is necessary to anticipate variations in the heat load. A constant cooling system heat load may be assumed since it will be a result of the construction methods employed and will be affected only slightly by changes in environmental or coolant temperatures within practical limits (approximately $\pm$ 40°K). The detector heat load, however, may change by a factor of two or more, depending on shifts in operational conditions with time. At the lower temperatures (solid hydrogen), a variation factor of two to three would cause a temperature change of only about 0.5°K, because of the large variation that can be tolerated. At the higher temperatures (solid methane) variations in detector heat load will have a much greater effect on the coolant temperature. However, if the orifice is manually variable (by means of a valve for example) a constant temperature could be achieved by periodic monitoring and valve adjustment over short periods of time when the detector heat load is constant.

The advantage of this system is its simplicity, since either a fixed or variable orifice can be employed and, once adjusted, no external power is required. The main disadvantage is that under conditions of varying heat loads, a fixed orifice device would result in temperature variations.

III. SELECTION OF MATERIALS AND FABRICATION OF MODEL K6 COOLER

The properties of the various coolants, insulation materials, and fabrication materials as well as evaluation of critical application techniques were closely correlated with the findings (Reference 1) of a previous intensive cryogenic study program.

The theoretical studies, calculations, and experiments described in previous sections of this report formed the basis for the design of Cryogenic-Solid Cooler Model K6 (Figure 10) for use with solidified methane. Each major assembly or component of the cooler is described in the following sections.

A. COOLANT CONTAINER

Figure 11 is a photograph showing the coolant container with one of the two dome-shaped ends welded in place. The cylindrical section of the container is made of 0.040-in. stainless steel (300 Series). This material was rolled into a cylindrical shape measuring 14 in. in length and 14 in. in diameter, and then welded. The two dome-shaped ends will add an additional 2 in. each to the length of the coolant container. These ends are made of .004-in. thick stainless steel (300 Series). This material provides the best strength-to-weight ratio and lends itself to attachments and sealing by means of soft and silver-soldering techniques.

B. OUTER CONTAINER

The outer container also shown in Figure 11 serves as a vacuum jacket around the coolant container. It is made of rolled 0.1875-in. thick aluminum for

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Reference (1): Investigation of Cryogenic-Solid Cooling Techniques, ASD-TDR-62-195, Aerojet Report No. 2127, February 1962.

maximum strength at the minimum weight. As in the fabrication of the coolant container, a sheet of the selected material was rolled into a cylinder 16-3/8-in. in diameter and 19 in. high, and welded to serve as the outer container. The end caps, also made of the same stock, were then formed into dome-shaped discs to withstand the atmospheric pressure.

The upper end-cap has provisions for mounting devices for evacuating the vacuum jacket, filling with coolant, venting, and for securing a pressure relief valve. The bottom end-cap has one large opening to accommodate the detector configuration. Both end-caps are welded to the cylinder. Flanges and connections on the end-caps are made of aluminum and are also welded in place.

C. HEAT TRANSFER ROD AND DETECTOR HOLDER

The heat transfer rod conducts heat from the detector to be cooled to the coolant in the container. Figure 10 shows the location of the heat transfer device which consists of a 1/2-in. diameter copper rod 20 in. long. The cold-finger end of this rod protrudes 2 in. through the end-cap, thereby making accessible the mechanism for positioning the detector. The opposite end of the rod is attached to a spring, which is anchored to the closed end of the container. This is done to compensate for thermal expansion. The cold finger contains a small chamber for the helium gas, used to monitor the temperature of the cold finger through the gas thermometer system. The rod is silver-soldered at all points of contact with the coolant container and the spring.

D. SUPPORTS FOR COOLANT CONTAINER

The coolant container is supported within the outer container by lines made of untreated Dacron webbing having a 50-lb. tensile strength. These are

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attached to the coolant container wall just below the end-cap welds (see Figure 12). Small stainless steel loops are placed 120 degrees apart around the circumference of the coolant container at each end to provide geometrically balanced suspension. At each point, two Dacron lines are taken axially and two radially from the loops to pulleys. The axial lines go through pulleys in the end-caps of the outer container. The radial lines go through pulleys on the wall of the outer container. The lines are then brought out in such a manner as to permit them to be tightened and fastened. This technique results in proper and rigid positioning of the coolant container.

With three anchor points at each end of the coolant container, the load is evenly distributed over a total of 24 Dacron lines.

E. FILL AND VENT PORTS

A port, Figure 13, is provided for filling the coolant container with the liquid helium. It may also be used to monitor the system pressure if desired. When not in actual use, this port should be sealed to prevent gas leakage into the system. The fill port consists of a 4-ft. length (approximately) of 0.250-in. O.D. stainless steel tubing having a 0.006-in. thick wall. It is silver-soldered to the coolant container and brought out in a long curved path to add thermal resistance to the outer container. It passes through the wall of the outer container via a jacket for greater thermal efficiency.

A separate vent port is provided for pumping over the liquid to bring about solidification of the liquid gas. This 4-ft. length of tubing is made of 3/4-in. O.D. stainless steel having a 0.010-in. thick wall. As with the fill port, the vent port tubing describes a long curve in going from the coolant container to the outer container. The vent port is silver-soldered at the point of entry into the coolant container, wherein it is attached to the pressure control valve (see Figures 8, 10 and 13).

F. PRESSURE CONTROL VALVE

The pressure control valve, Figure 8, whose operation was described in Section IIC, is made of stainless steel parts and employs a brass bellows. It is actuated by pressure changes in the helium gas contained in the cold finger chamber. Two 0.042-in. O.D. x 0.006-in. thick wall stainless steel tubes are attached to this device for purging and filling the helium gas system and for transmitting pressure changes from the cold finger to the bellows within the pressure control valve.

G. INSULATION

Although the coolant container is surrounded by a vacuum, heat will be transferred to it by radiation. In order to minimize the effects of such radiation, layers of insulation are wrapped around the coolant container (Figure 14). This insulation consists of reflective shields of aluminized Mylar (Figure 15) separated by a cloth insulator-spacer. There are 90 layers of shielding and spacers for every 3/8-in. of insulation thickness. This insulation is located in the evacuated areas separating the coolant and outer containers. All tubes and wires which penetrate the insulation (Figure 10) are brought out through a few layers at any one location, thereby giving the component additional radiation shielding.

The Dacron lines, of necessity, penetrate the insulation directly; however, these do not contribute appreciably to either the conducted or radiated heat arriving at the coolant container.

IV. RESULTS OF TEST ON UNMODIFIED COOLER

The operational life of the system was determined after the insulated coolant container was installed in the outer container and the insulated space between the containers was evacuated. The completed cooler assembly is shown in Figure 1. The cooler was charged with 40 lb of methane which was subsequently solidified in preparation for the gas flow-rate test which would serve as the basis for calculating the results of the operational life.

A. GAS FLOW-RATE TEST

The purpose of this test was to determine the rate at which the coolant would sublime as a result of the system heat load minus the detector heat load. The following data was recorded during the six-day test:

Date	Av Rate of Gas-Flow/Day (ft^3/min)
2-2-65	0.0036
2-3-65	0.0021
2-4-65	0.0024
2-5-65	0.0046
2-6-65	0.0025
2-7-65	0.0028
2-8-65	0.0029
Total =	0.0209
Average for seven days =	0.00299 ft^3/min

B. HEAT LOAD OF OPERATING SYSTEM

In order to determine the heat load

$$Q = \frac{\text{heat of sublimation/ft}^3\ \text{methane/min}}{1\ \text{watt}} \times (\text{Av. gas flow rate})$$

1 watt = 14.34 cal/min.

lb methane = 24.04 ft^3 gas.

1 lb methane = 61,644 cal.

Sublimation rate of 1 ft^3 methane/min = 2564 cal/min,

therefore:

$$Q = \frac{2564}{14.34}(0.00299)$$

$$= (178.8)(0.00299)$$

$$= 0.535 \text{ watt}$$

C. ANTICIPATED OPERATIONAL LIFE OF COOLER

The operational life of the cooler was calculated as

$$T = \frac{W_c H_s}{KQ}$$

where:

T = operating time in mo.

W_c = coolant fill in lb.

H_s = heat of sublimation in cal/lb.

Q = heat load, 0.535 watt.

K = a constant, 6.28×10^5.

therefore

$$T = \frac{40(61644)}{6.28 \times 10^5 (0.535)}$$

$$= 7.34 \text{ months}$$

D. SYSTEM TEMPERATURE

The temperature of the system at the cold finger was recorded with and without the detector load with the following results:

1. Without the detector load -
   Pressure over solidified methane: 2.2 Torr
   Temperature: 77.3°K

2. With detector load (0.1 watt) -
   Pressure over solidified methane: 2.1 Torr
   Temperature: 79.4°K

The results of this test illustrated the basic performance characteristics of the cryogenic-solid cooler. The operational life fell short of the calculated time, however, this is attributed to the conservative heat load values used in the calculations, particularly with respect to insulation. The actual heat load was approximately a factor of two greater than predicted. The desired temperatures, however, were achieved. The unit was subsequently modified to enable operation and use with instruments requiring an infrared detector.

V. MODIFICATION OF COOLER

A. IMMOBILIZATION OF PRESSURE CONTROL VALVE

Improper operation of the internal temperature-control valve, caused by blockage of the small diameter capillary line at cryogenic temperatures, made it necessary to immobilize the valve. The valve was reopened at ambient temperature by external pressurization. This enabled the evacuation of the valve bellows, capillary lines, and temperature sensing bulb followed by the injection of a filler under pressure. The filling material was Cerrolow 17 alloy selected for its low viscosity in the liquid state and low fusion temperature. After cooling, tests showed that the valve had been successfully immobilized in the open position. To ensure lasting immobilization, the capillary tube ends were capped.

B. DETECTOR MOUNTING ASSEMBLY

A flexible cold-finger detector assembly was designed and fabricated for use with the cooler. The insulation pad covering the cold-finger stub on the coolant container was removed and the flexible cold-finger detector assembly (without the detector) was attached. The flexible cold finger, encased with the same superinsulation material as the remainder of the cooler, was supported and centered in its frame with 0.010-in. dia. stainless steel wires. At the same time that the detector assembly was attached to the cooler, a getter (Linde 4A Molecular Sieve) was placed under the insulation to help maintain vacuum.

C. PRESSURE RELIEF SAFETY VALVE

A burst-type pressure relief valve is provided with the unit. This valve will open in the event that the pressure in the vacuum jacket exceeds 15 psi, thereby, preventing damage to the coolant container.

D. PORTABLE HANDLING DEVICE

A cart was fabricated to facilitate handling the system (Figure 2). This cart permits the cooler to be moved and positioned in any attitude.

VI. PERFORMANCE BY MODIFIED COOLER

Subsequent to the cooler modifications described in the previous section, the system was charged with 25 lb of liquid nitrogen, which is a volume equivalent of 14.1 liters. The gas flow-rate of the system was then measured and determined to be 0.00686 ft^3/min with a system heat load which included the flexible detector holder and 0.1 watt simulated detector. The 0.00686 ft^3/min. average flow-rate value recorded over the 15-hr. and 13-min. test duration was used in calculating the operating heat load and life. Prior to the flow-rate measurements, the charged cooler was allowed 74 hr. to attain thermal equillibrium.

The following is a summary of the characteristics of the modified K6 cooler. (Reference Section 4.5.4 test data of JPL Spec. No. GMY-15060-DSN-A).

1.	Time to reach thermal equilibrium	24 hr
2.	Temperature	
	a. At detector mount (CH_4)	80 to 85.8°K
	b. In coolant chamber (CH_4)	68°K
	(See Section VIC for other coolants)	
3.	Temp and Temp Stability of the infrared detecting device simulator (CH_4)	80-85.8°K±0.5°K
4.	Rate of weight loss of the solid coolant	0.015 lb/hr
5.	Calculated operational life (CH_4)	5.28 mo.

6. Projected coolant life with:

LN_2 (77°K)	
SN_2 6.5°K	See Section VIC
SCO 50°K	

7. Pumping Interval — Continuous pumping to maintain a solid.

8. Max time interval permissible between pumpdown times (Insulation Area) — Every 6 mo.

9. Extrapolation of heat loads for various environmental temperatures — 350°K - 0.921 watts; 250°K - 0.600 watts

10. Effects of various ambient temperatures on the system life — 350°K - 4.27 mo.; 250°K - 6.55 mo.

A. OPERATIONAL HEAT LOAD

In determining the heat load as

$$Q = \frac{\text{heat of sublimation of } N_2/ft^3 \text{ at NTP}^*}{1 \text{ watt}} (\text{gas flow-rate}/ft^3/\text{min})$$

1 watt = 14.34 cal/min

1 lb.N_2 = 13.8 ft^3 of gas at NTP

1 lb.N_2 = 21,500 cal

The sublimation rate of N_2/ft^3 at NTP = 1556 cal/min,

therefore:

$$Q = \frac{1556}{14.34}(0.00686)$$
$$= 108.4\,(0.00686)$$
$$= 0.745 \text{ watts}$$

*Normal temperature and pressure

B. OPERATING LIFE

The operational life of the cooler charged with Solid Methane was determined as

$$t = \frac{W_c \, H_s}{BQ}$$

where:

W_c = 4016

H_s = 61,700 cal/lb

B = 619,000 cal/watt mo.

Q = .745 watts

$$t = \frac{3.926}{0.745}$$
$$= 5.28 \text{ months}$$

C. TEMPERATURE MEASUREMENTS

Two nickel resistance thermometers and one copper constantan thermocouple were attached to the screw-on tip of the detector assembly for monitoring the temperature.

The two nickel resistance thermometers, Arthur C. Ruge Associates, Inc. part No. BN-200, were epoxied to the detector holder.

The calibration chart for a BN-200 nickel resistance thermometer is shown in Figure 16. This represents resistance measurements made with a Wheatstone Bridge at liquid nitrogen temperature (77.4°K), liquid argon temperature (83.7°K), and Freon 14 (145°K). The two resistance thermometers were found to be 21.7 and 22.6 ohms, respectively, at liquid nitrogen temperature.

With liquid nitrogen in the cooler, the resistance thermometers measured 28.3 and 29.0 ohms, respectively. From Figure 16, temperatures of 89.4 and 90.5°K, respectively, are determined. This gives temperature gradients across the detector assembly of:

$$89.4 - 77.4 = 12^{o}K$$
$$90.5 - 77.4 = 13.1^{o}K$$

The copper-constantan thermocouple's emf was 0.282 mv. referenced against liquid nitrogen. This is equivalent to a temperature gradient of $15.8^{o}K$.

It is not expected that the anticipated increase in the ΔT of $9.4^{o}K$, of solid methane @ $68^{o}K$ when compared to Liquid N_2 ($77.4^{o}K$) will add to the heat load.

The heat transfer modes present in Aerojet superinsulation are conduction, which varies linearly as the temperature, and radiation which varies as the fourth power. It has been noted that the apparent thermal conductivity of superinsulation decreases as the cold boundary temperature is lowered. It was determined that the heat leakage may even decrease with the lowering of the cold boundary temperature.

Assuming a heat leak of 0.745 watts and an expected life of 5.28 months, based on the test results, the expected life and coolant temperature for other liquid and solid coolants is given by:

$$(T_{v_1 S}) = \frac{(5.28)\ (\rho_{v_1 S})\ (H_{v_1 S})}{(0.522)\ \ (136)} = 0.0745\ (\rho_{v_1 S_1})\ (H_{v_1 S})$$

in which $T_{v_1 S}$ is the operating time in months for the new liquid or solid coolant in months,

$\rho_{v_1 S}$ is the density of the new liquid or solid coolant in gms/cm^3,

$H_{v_1 S}$ is the heat of vaporization or sublimation of the new liquid or solid coolant in cal/gm.

Coolant	ρ_{v_1S}	H_{v_1S}	T_{v_1S}	Temperature, ^oK
LHe	0.125	4.90	0.0456	4.2
LH_2	0.071	107	0.566	20.4
LNe	1.21	20.7	1.87	27.2
LN_2	0.808	47.6	2.87	77.0
LCO	0.803	5.1	3.06	81.6
LA	1.4	39.1	4.07	87.4
LO_2	1.14	50.8	4.32	90.1
LCH	0.425	122	3.77	111.7
SH_2	0.0808	120.9	0.728	10
SNe	1.443	24.7	2.66	16
SN_2	1.02	53.7	4.08	47
SCO	1.02	58.0	4.40	51
SA	1.654	45.8	5.65	67.

In most cases, the 45-lb stress limit of the K6 coolant container suspension system will not allow full capacity for the above alternate coolants. In addition, the temperature gradients across the detector assembly must be added to the above temperatures.

VII. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

The validity of the basic cryogenic-solid cooling system design has been demonstrated. Although the estimated heat loads proved to be conservative resulting in a shorter operating life (5.28 mo.) than desired, the temperature measured at the detector mount was lower than anticipated. The resultant expected life is within the limits stipulated in the specifications.

The performance with methane instead of the nitrogen used in the test will fulfill all requirements.

The addition of the flexible detector holder assembly increased the heat load from 0.535 watts to 0.745 watts. The ΔT between the coolant and cold finger end was determined to range between 12 and 15.8°K.

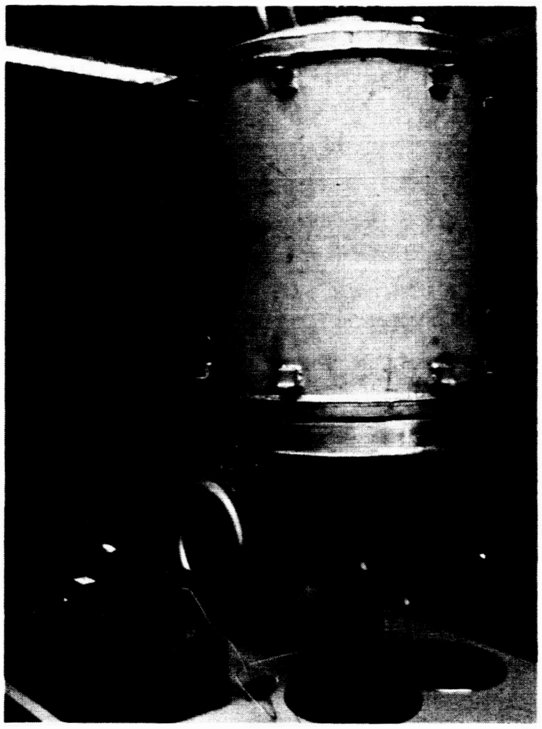
For this test, liquid nitrogen having a normal boiling point of 77.4°K was used, however, the cooler will actually use solidified methane at approximately 68°K. As a result, the expected temperature will be 68°K plus between 12 to 15.8°K resulting in a total temperature of between 80 and 85.6°K.

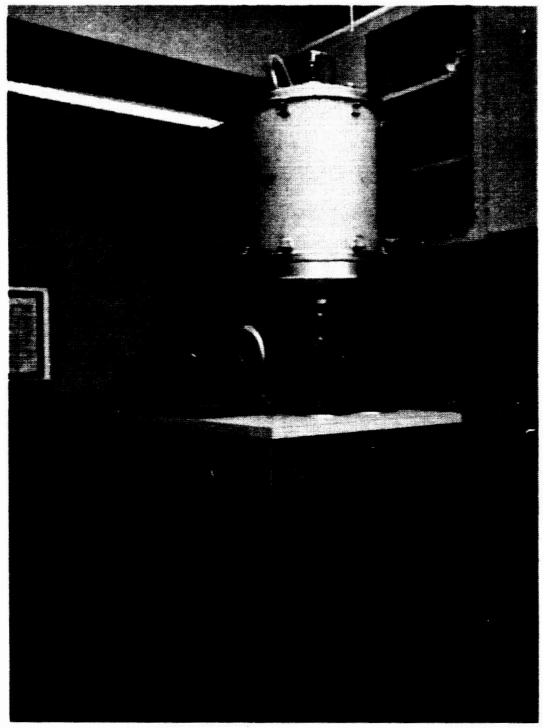
B. RECOMMENDATIONS

The following areas should be investigated further for methods of increasing the operating life of the solid cooler:

1. Minimization of heat in leakage by:
   a. Use of additional insulation
   b. Improving insulating techniques
   c. Achieving greater coolant capacity for the same system weight by optimization of system design, i.e., elimination of outer casing in space environment
2. Recovery of refrigeration in vented vapor to establish more desirable gradient in insulation.
3. Passive and shielding techniques tailored to specific mission requirements.

Methods of improving the solid cooler in the above and other areas are expected to result from the study phase of this program that is currently in progress.

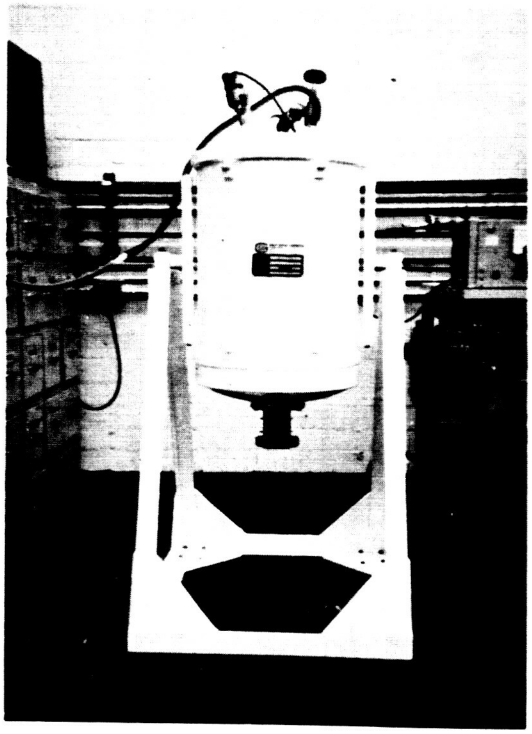




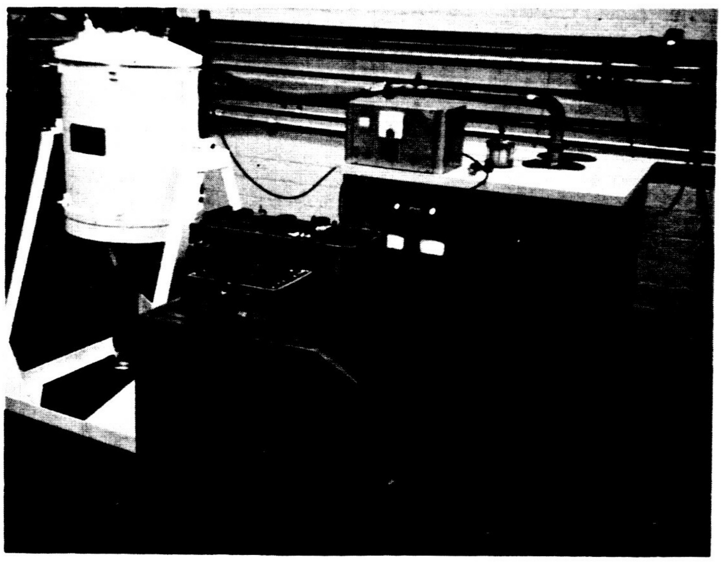
HEAT LEAK TEST SET UP OF CRYOGENIC-SOLID COOLER MODEL K6

A3813:65-1234

Figure 1



A. FRONT VIEW OF COOLER AND PORTABLE CART



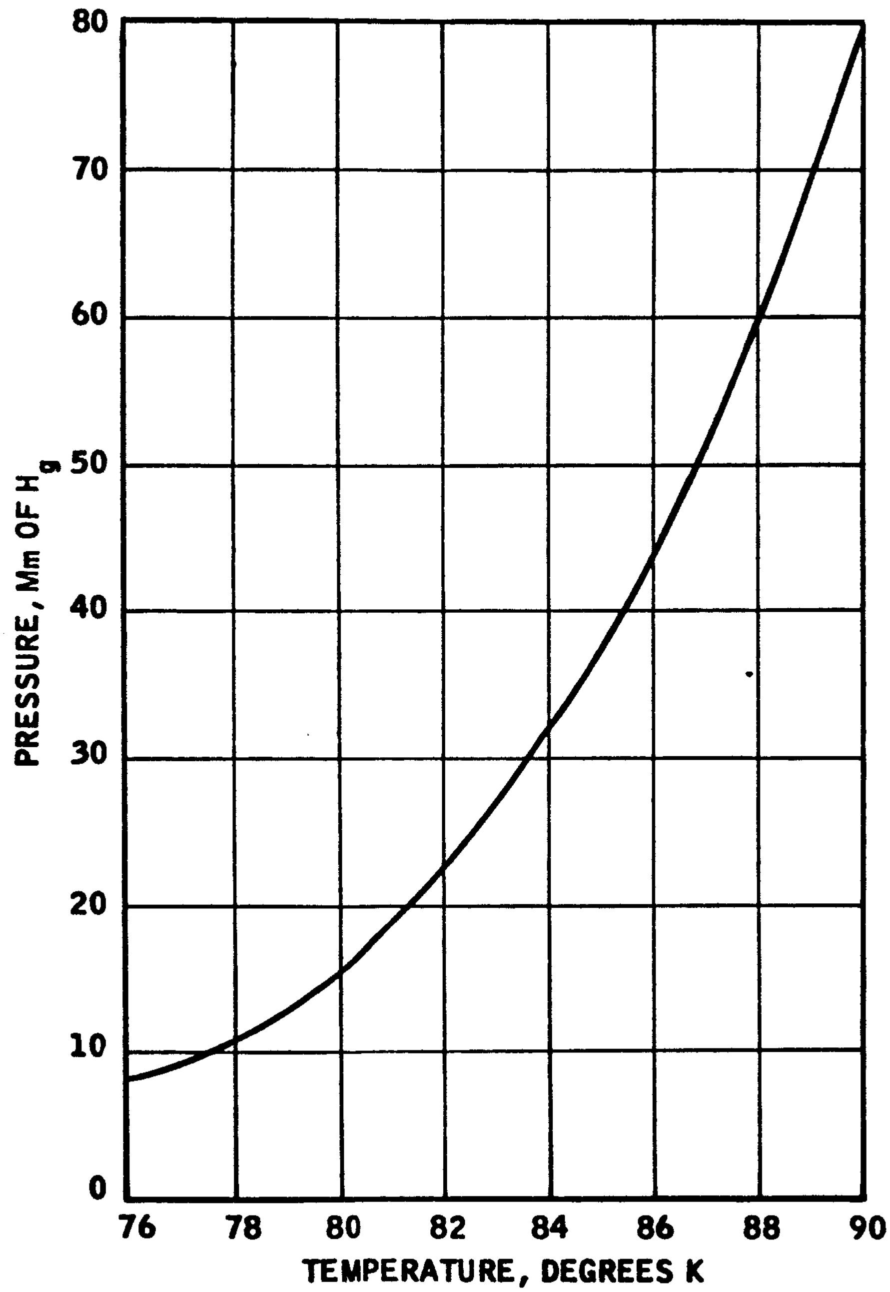
B. COOLER CONNECTED TO TEMPERATURE AND VACUUM JACKET PRESSURE MONITORING INSTRUMENTATION

MODIFIED CRYOGENIC-SOLID COOLER MODEL K6
AND PORTABLE CART

AEROJET GENERAL
ASTRIONICS DIVISION

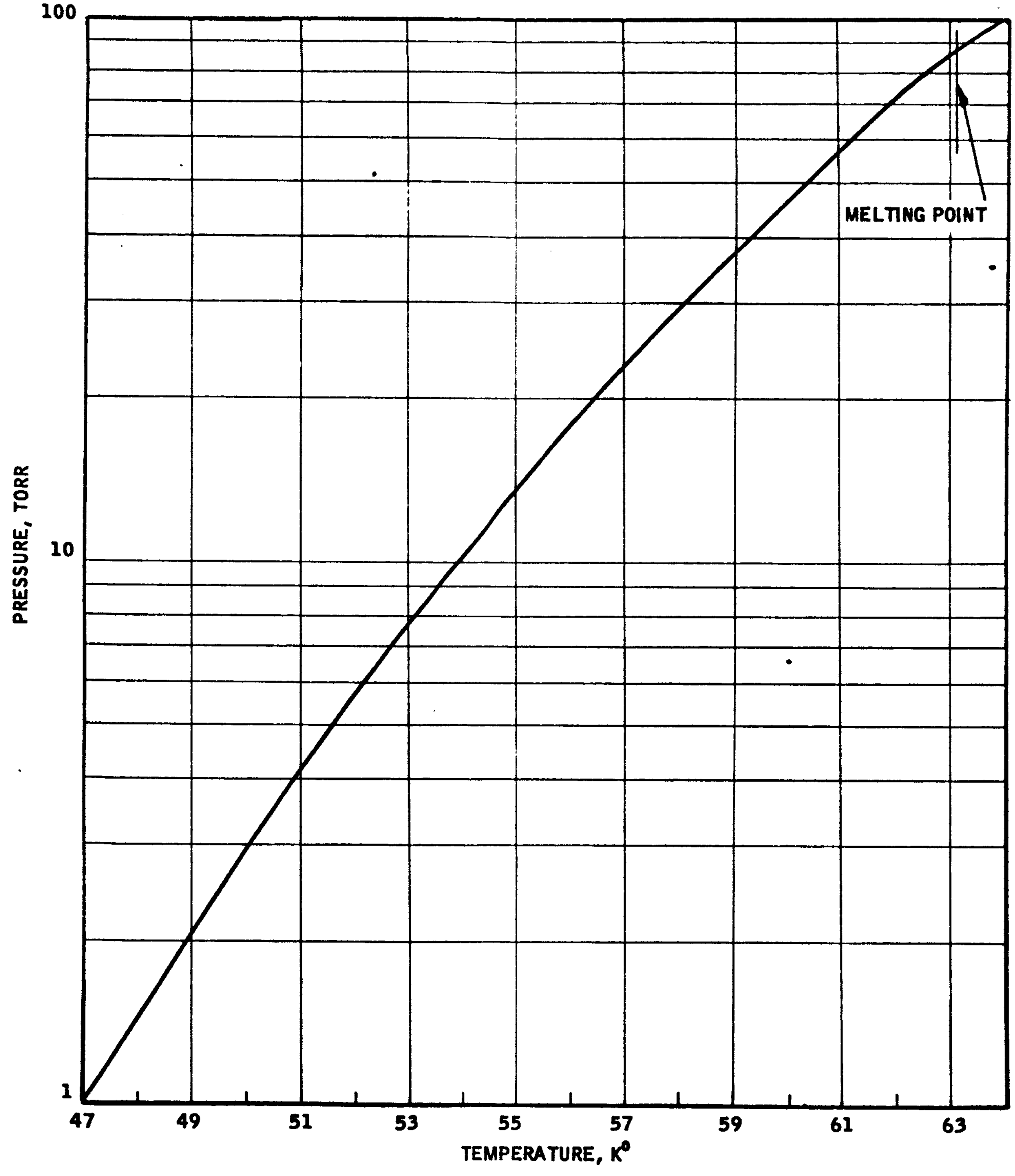
A3813:65-1253

Figure 2



VAPOR PRESSURE OF SOLID METHANE

A529:61-1083A



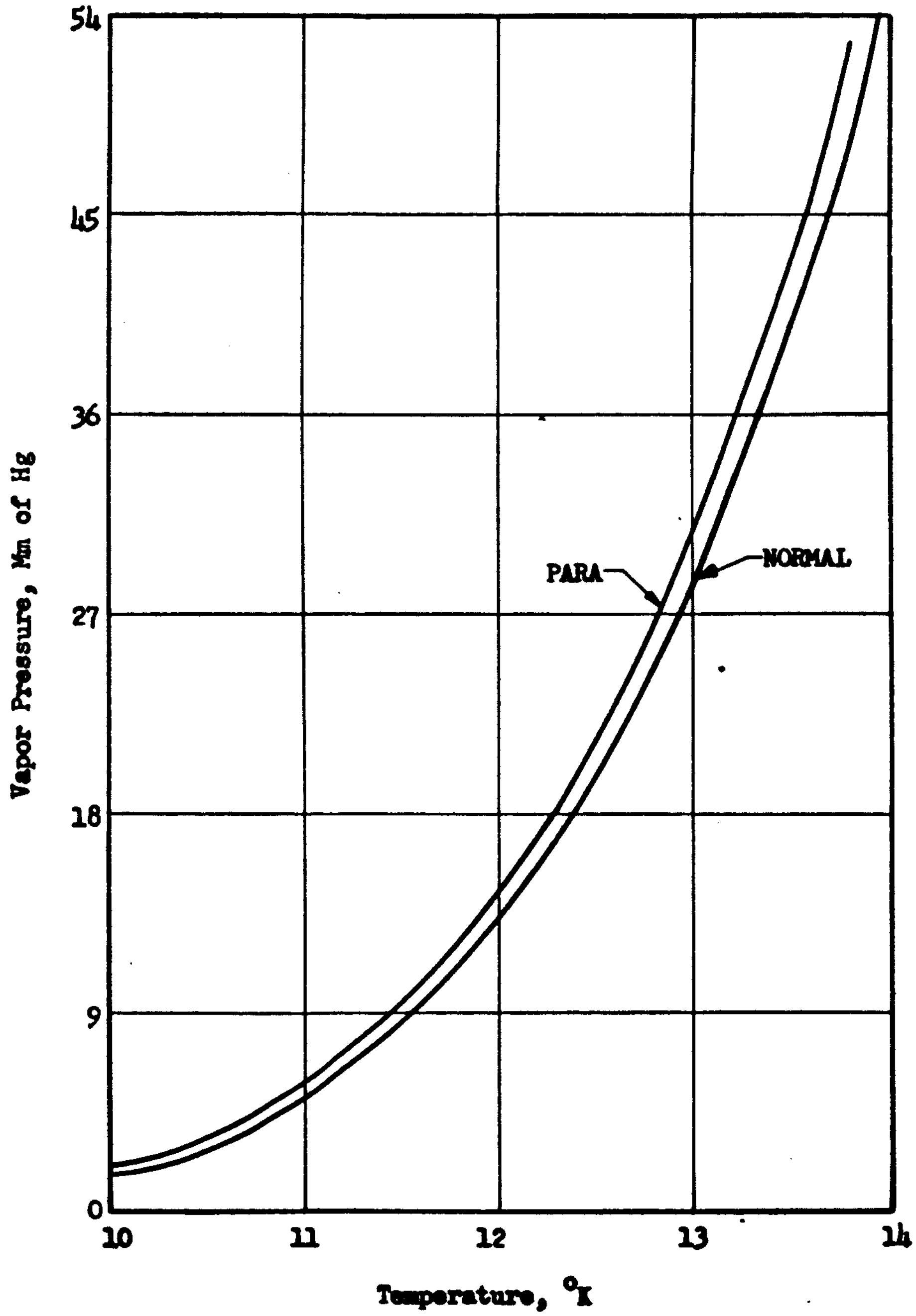
SOLID NITROGEN AS A FUNCTION OF VAPOR PRESSURE

B3813:65-1233

Figure 4

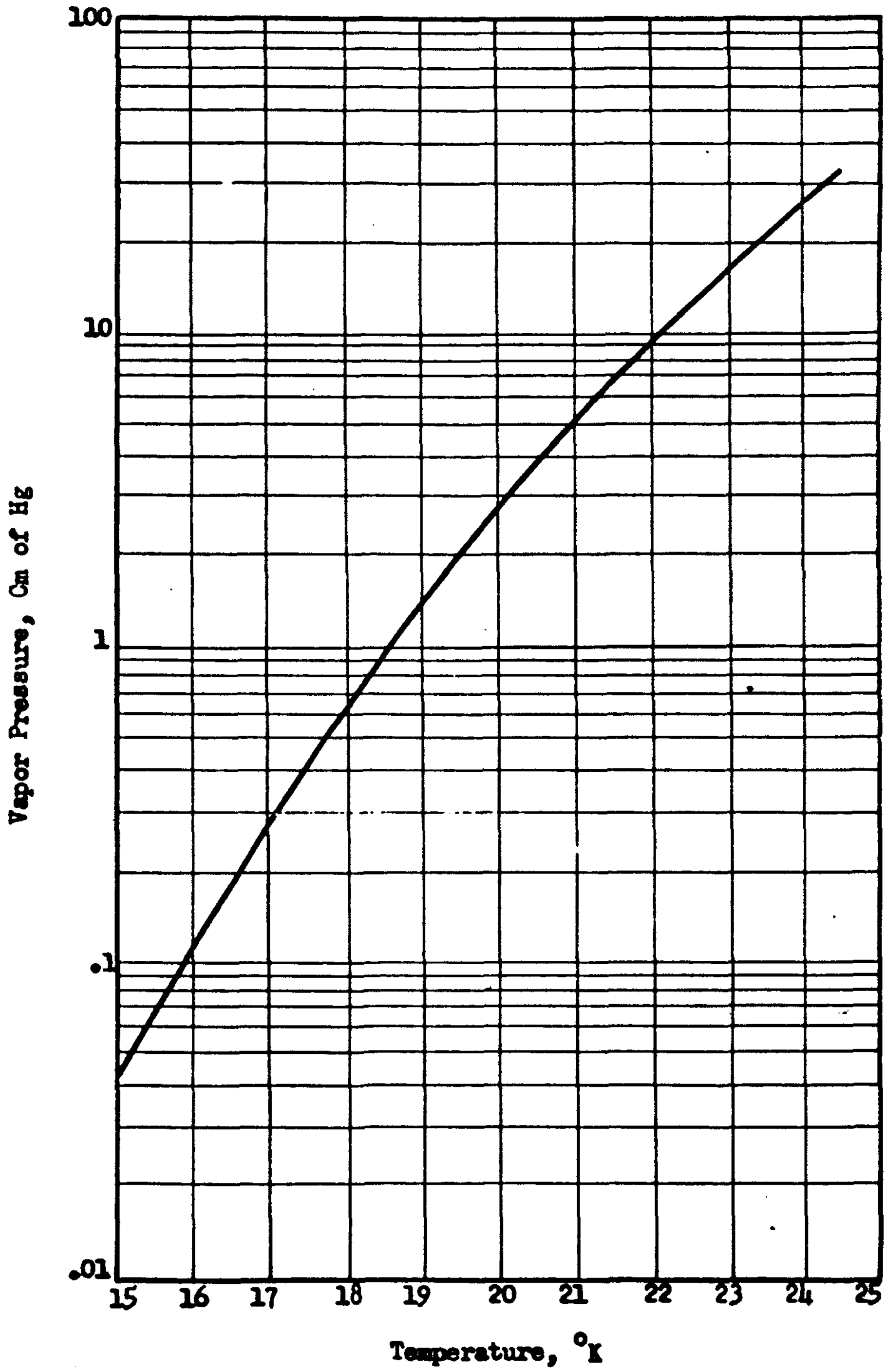
AGC 96-8

Aerojet-General Corporation
Azusa, California



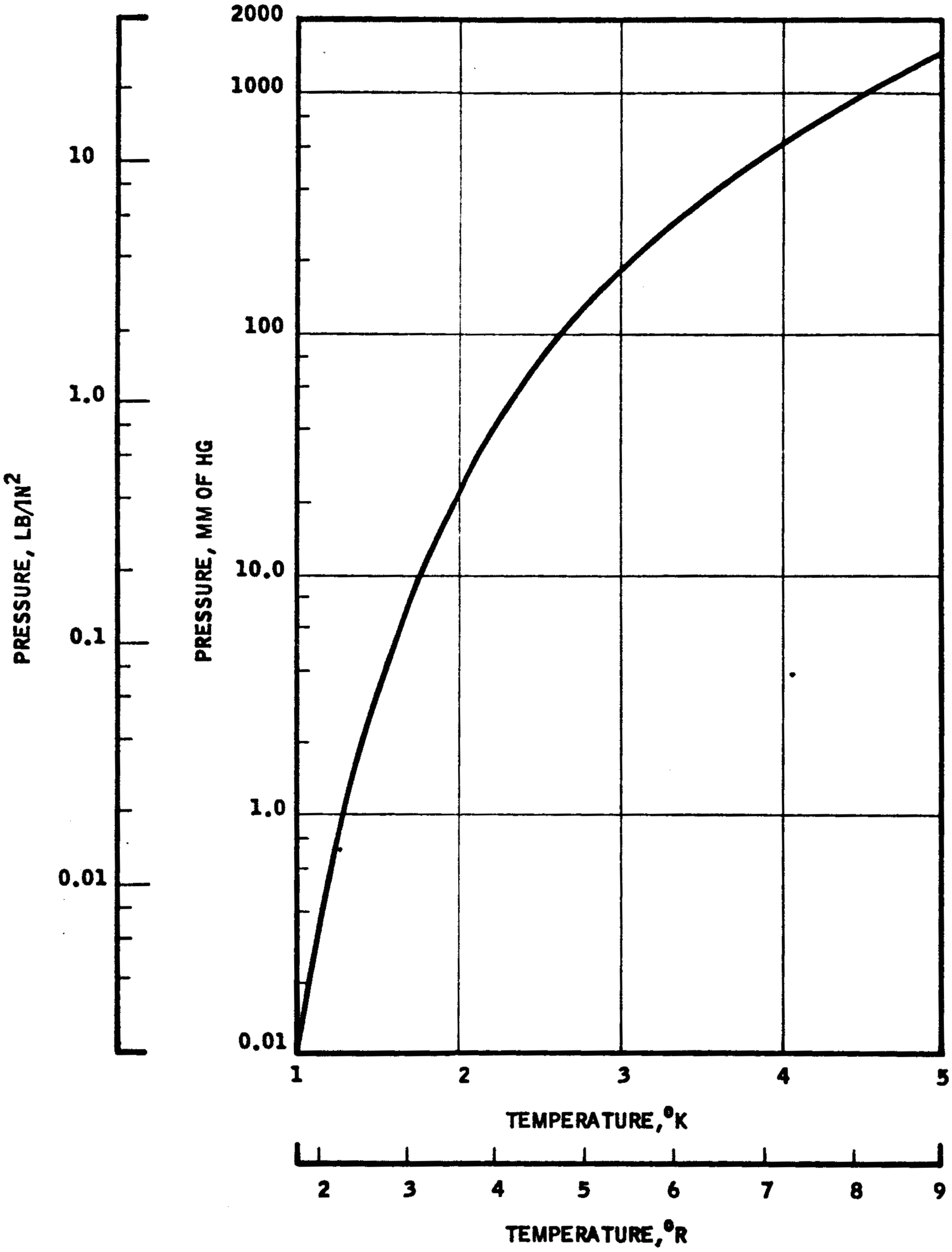
VAPOR PRESSURE OF SOLID HYDROGEN
(NORMAL AND PARA)

A529:61-1081



VAPOR PRESSURE OF SOLID NEON

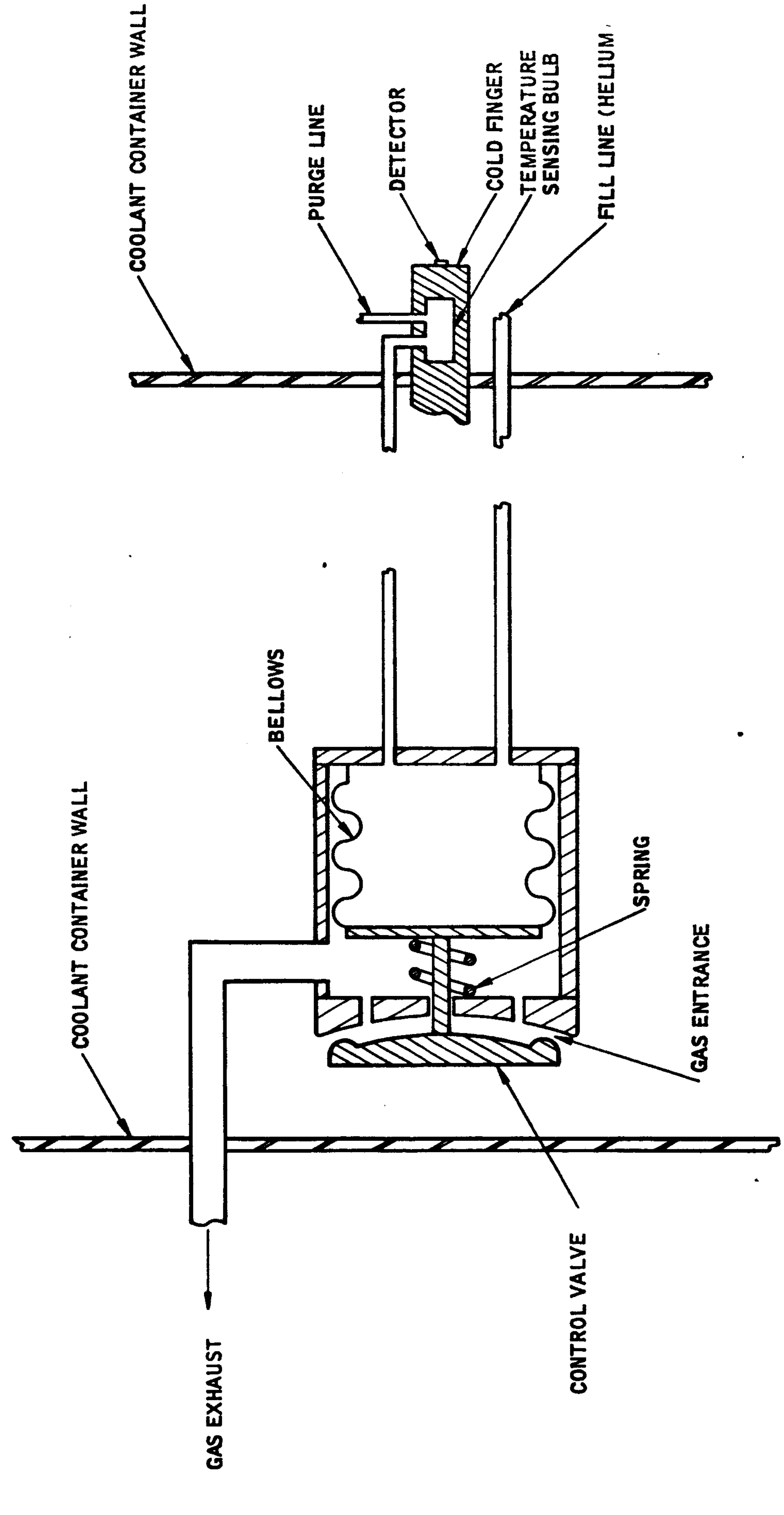
A529:61-1078



VAPOR PRESSURE OF LIQUID HELIUM

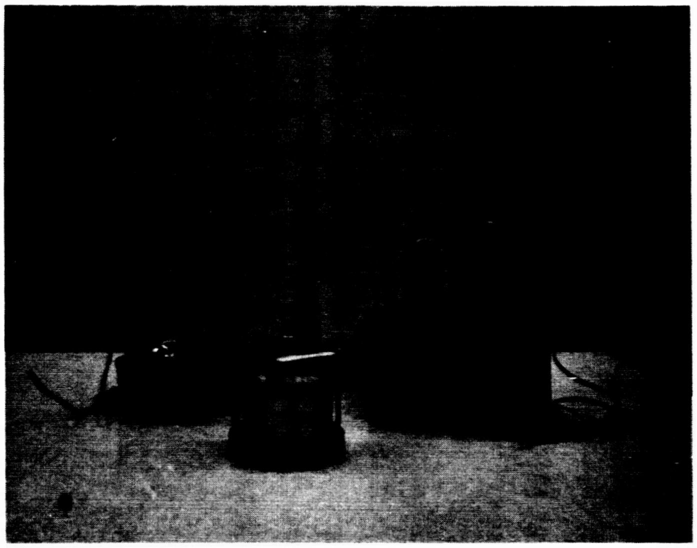
B3813:65-1227

B529: 64-1316



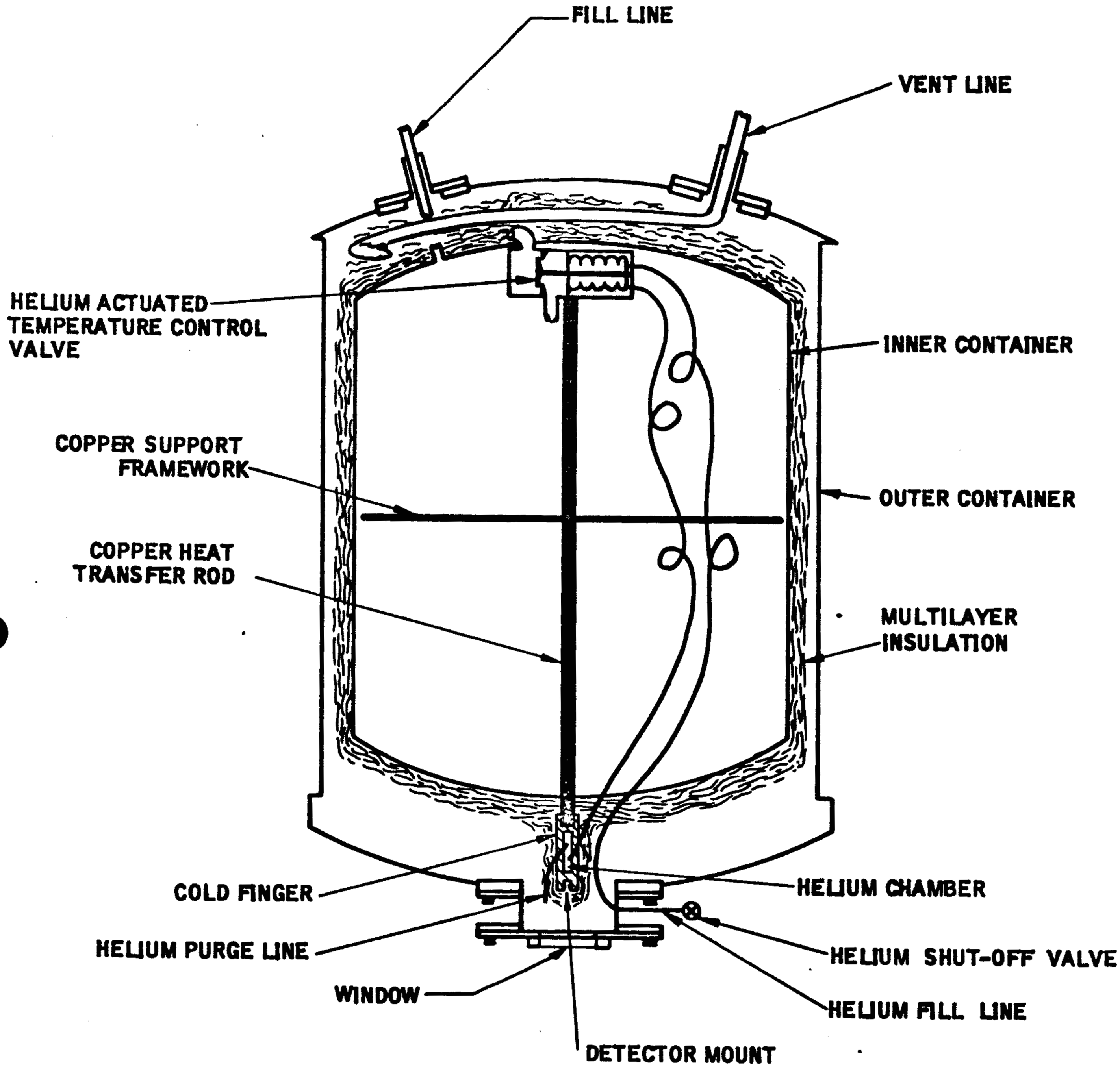
TEMPERATURE CONTROLLED PRESSURE RELIEF VALVE

Figure 8



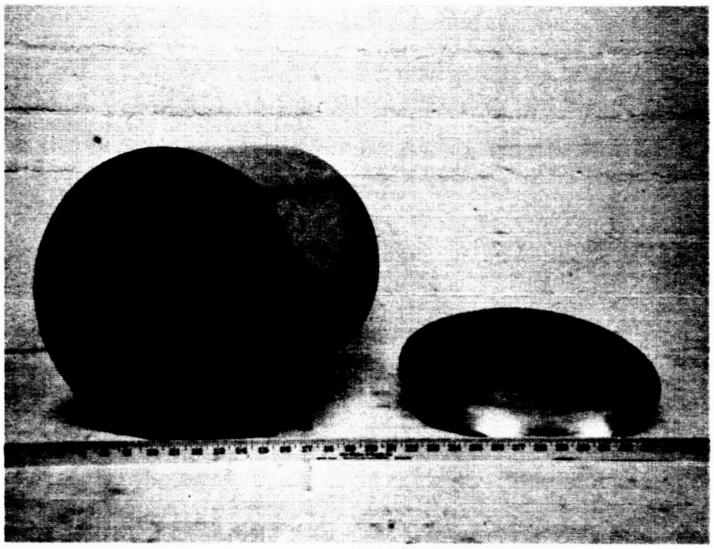
PRESSURE CONTACTOR RELAY ASSEMBLY AND VALVE

A3813:65-1235

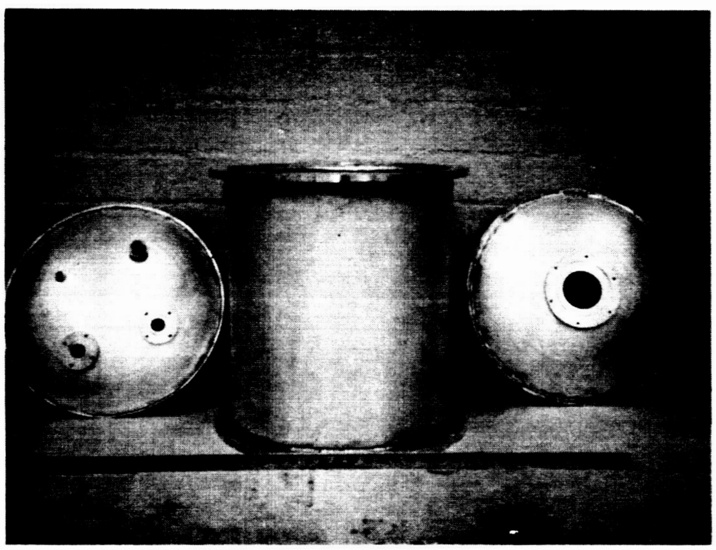


SOLID COOLER DIAGRAM

B529: 64-1318A



A. COOLANT CONTAINER AND END CAP

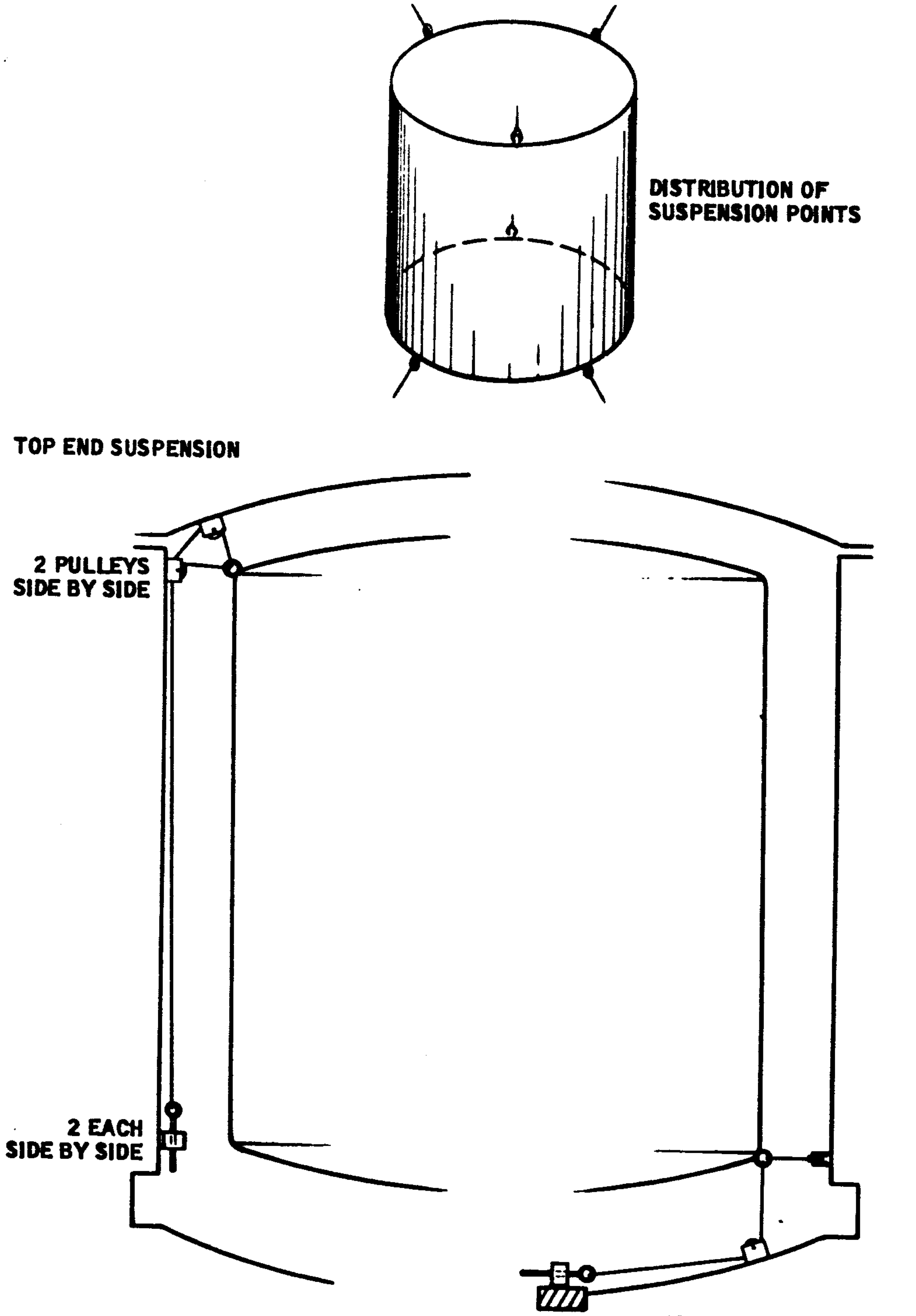


B. OUTER CONTAINER WITH TOP AND BOTTOM END CAPS, FROM LEFT TO RIGHT

COOLANT AND OUTER CONTAINER ASSEMBLIES

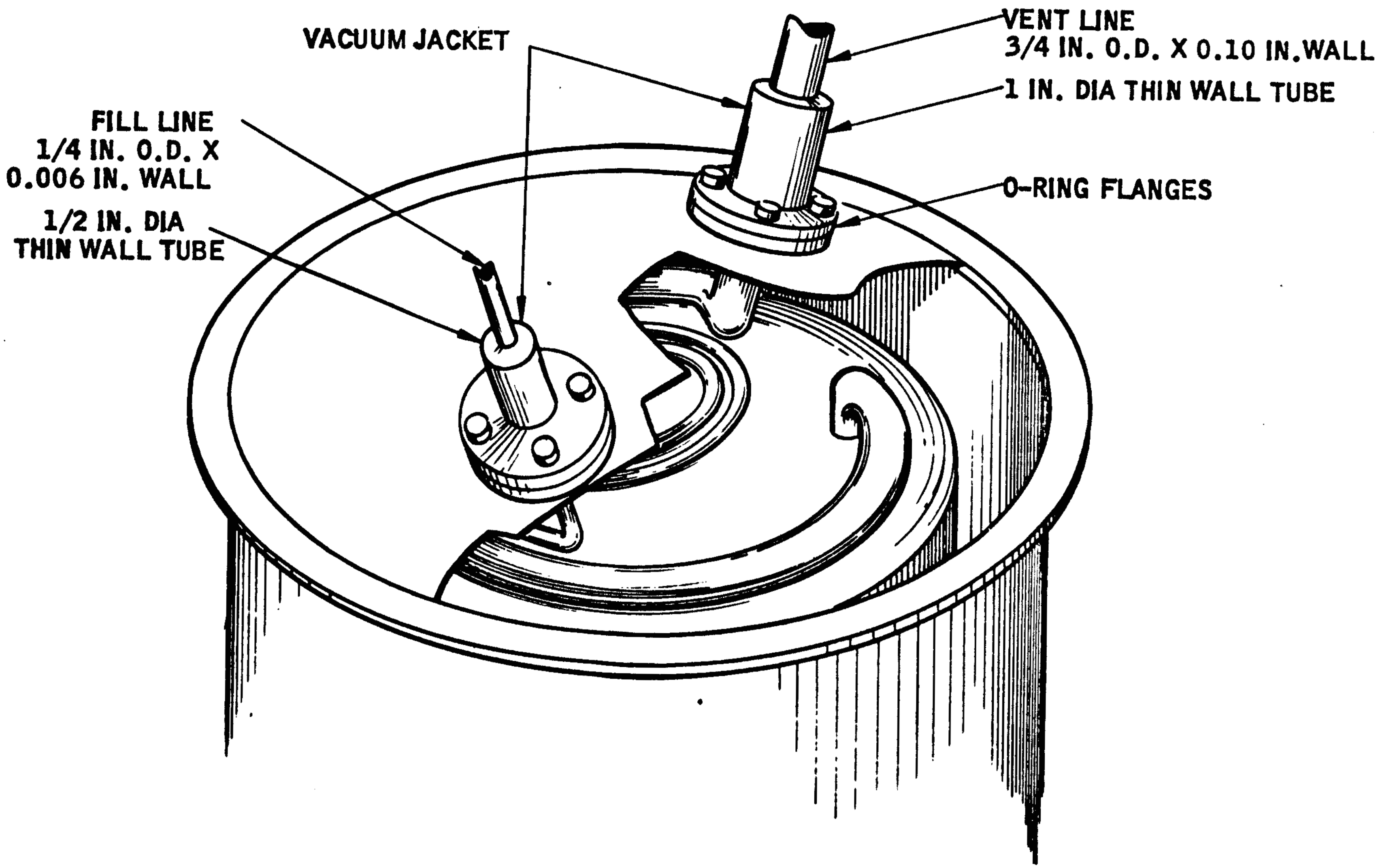
A3813:65-1236

Figure 11



SUPPORT SYSTEM FOR INNER CONTAINER

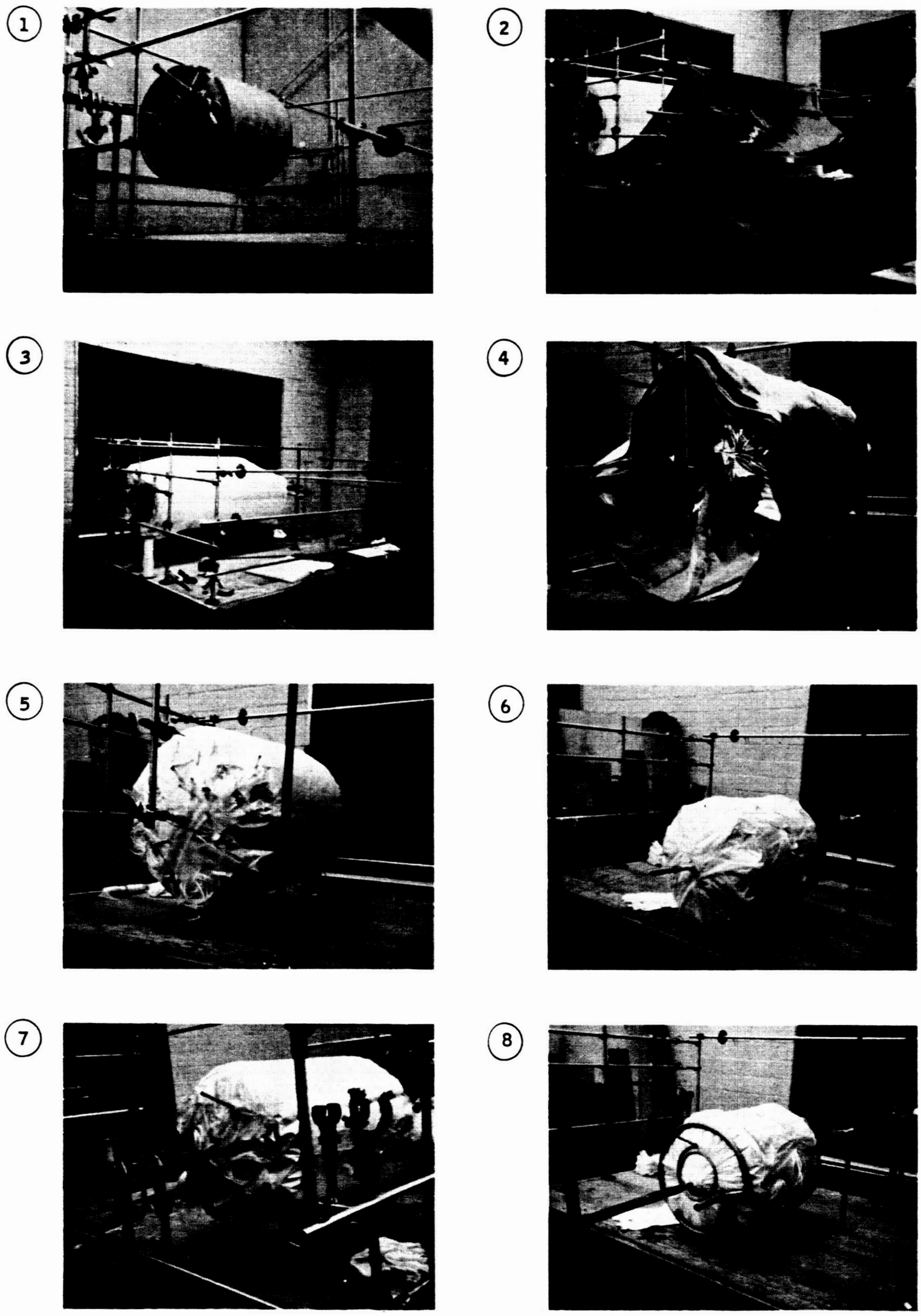
B529: 64-1314



CONFIGURATION OF FILL AND VENT PORTS

A529: 64-1315

Figure 13

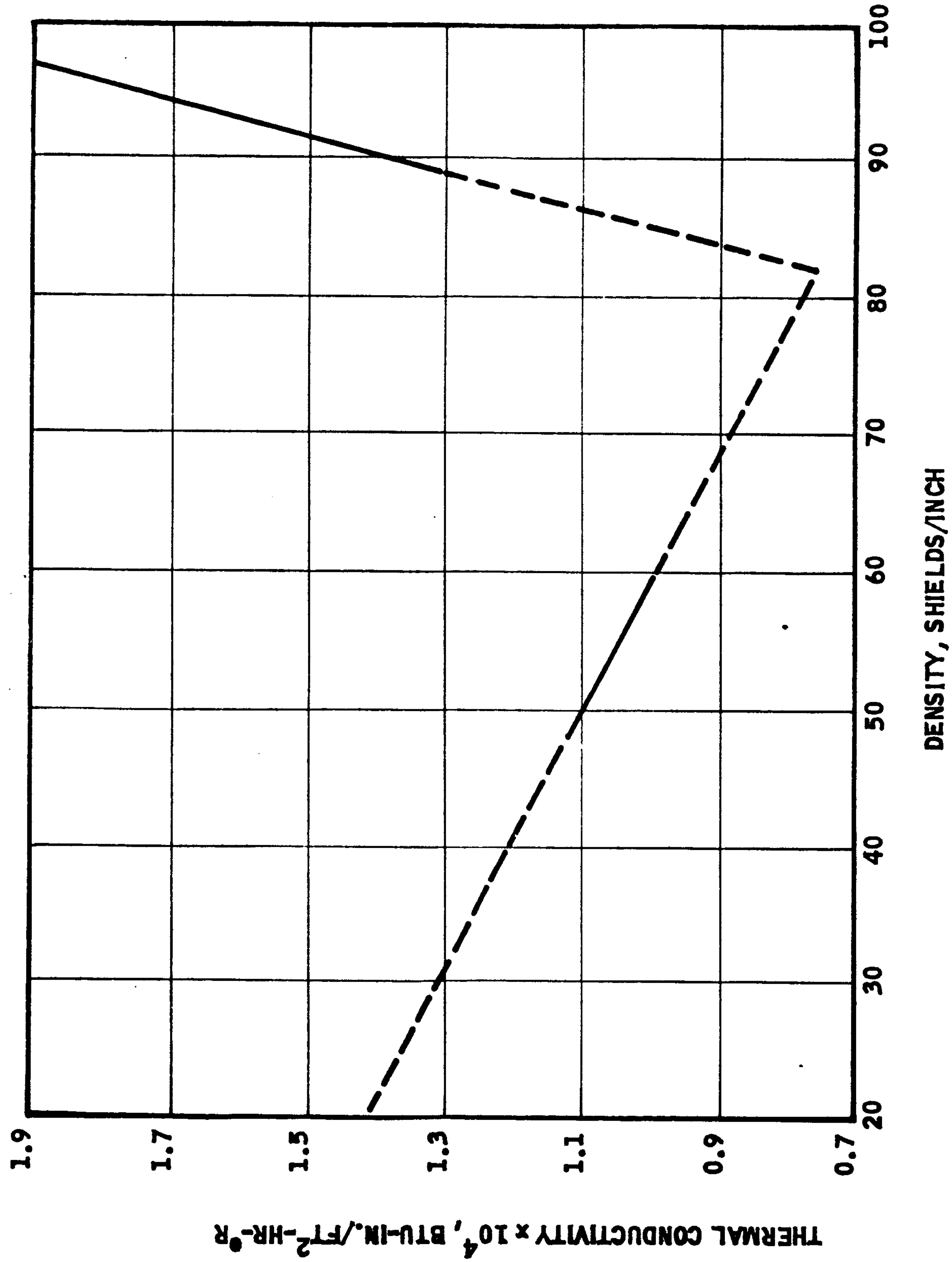


INSULATION APPLICATION PROCEDURE

A3813:65-1237

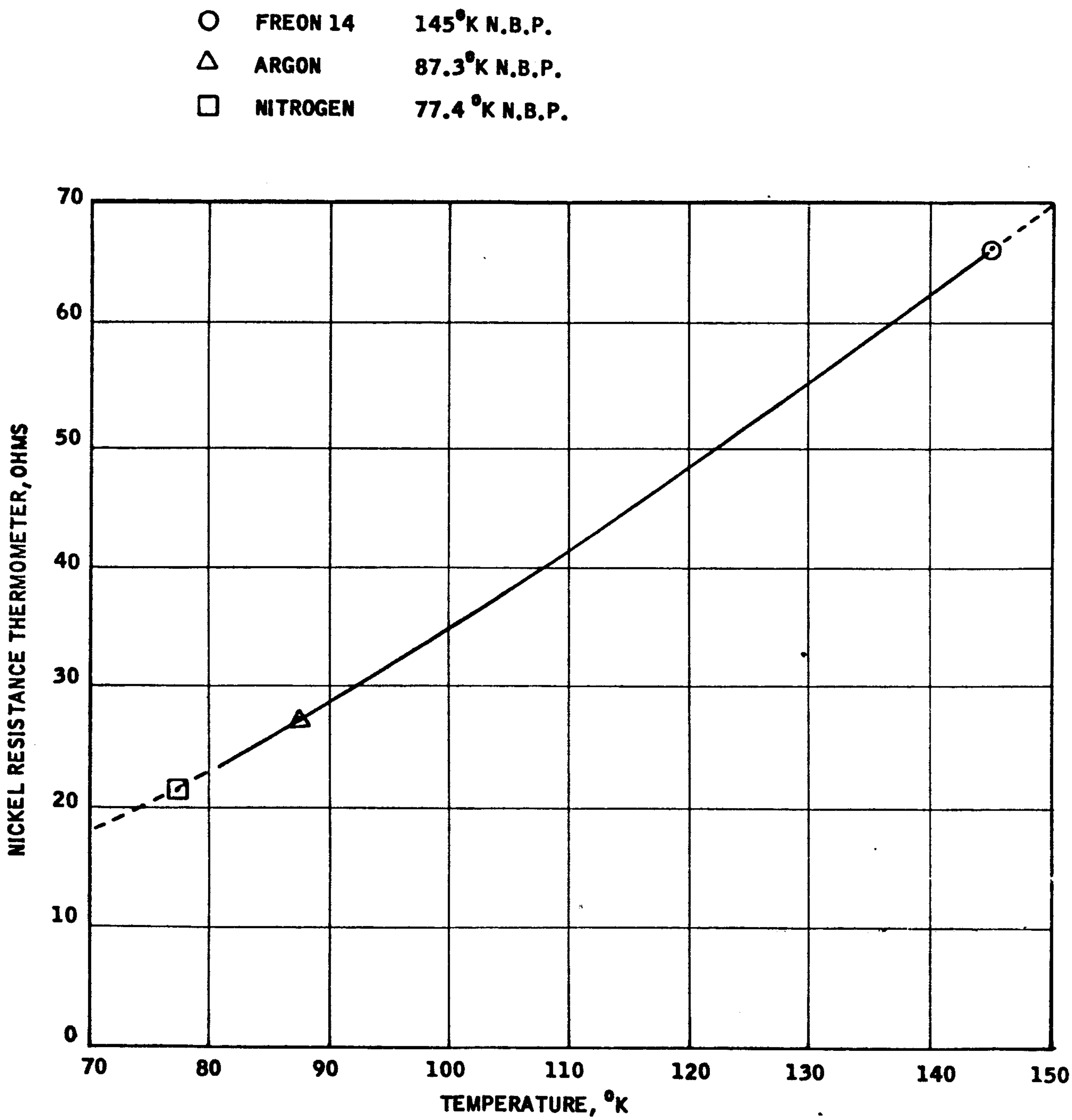
Figure 14

A3815:65-433



SUPERINSULATION
ALUMINUMIZED BOTH SIDES OF MYLAR

Figure 15



CALIBRATION CURVE FOR NICKEL RESISTANCE THERMOMETER

APPENDIX A

INSTRUCTIONS FOR OPERATING CRYOGENIC-SOLID COOLER MODEL K6

The following procedure will enable the safe and efficient preparation of the cooler for ground operation.

I. MOUNTING COMPONENTS ON COLD FINGER

Mount the IR detector or other component to be cooled on the cold finger as follows:

A. Break the vacuum, if present, with dry nitrogen (preferably from vaporized liquid nitrogen to prevent contamination).

B. Remove cover plate and housing while dry nitrogen purge is maintained through insulated space between the coolant container and the outer container.

C. Remove insulation pod from cold finger end of cooler.

D. Unscrew end of cold finger rod by turning counter-clockwise.

E. Solder component to end of cold finger and reinsert cold finger rod into threaded socket. Do not apply excessive torque, otherwise, it could result in the stainless steel detector mount supports being sheared off.

F. Reinsulate the rod and detector using Aerojet-General super-insulation or its equivalent. Care must be exercised to minimize extraneous radiation upon the cold finger end and the detector. Extra care must be exercised in applying the super-insulation at this point to avoid unnecessarily high heat losses.

G. Connect electrical component leads to spare terminals of hermetically sealed feedthrough mounted on housing.

H. Replace housing and new cover plate making certain that O-ring is in usable condition, as well as clean. Coat O-ring with good quality vacuum grease.

A 1/32-in. dia. hole was drilled 1/2 in. from the exposed end of the cold finger. It goes from the outer diameter of the rod to the inner thread and serves as a receptacle for a small temperature-monitoring thermocouple used during testing. The thermocouple is secured in place by means of a short 1/4 - 28 set screw. This set screw must not interfere with the threaded (male) portion of the component to be cooled so that the back surface of the component can make physical contact with the flat surface at the front of the cold finger. This contact is necessary to assure optimum thermal continuity between the component and cooler.

II. EVACUATION OF INSULATION SPACE BETWEEN INNER AND OUTER CONTAINERS

The space between the coolant and outer container should be evacuated as follows for maximum thermal isolation:

A. Attach the container to a high-vacuum pumping station with the Richards seal-off valve handle provided.

B. The Richards valve should still be closed. Rough out (to approximately 0.01 Torr) all accessory lines.

C. Push the Richards valve in to mate with the valve seat fitting. Open the valve by rotating the handle counter-clockwise. Rough out the container to approximately 0.01 Torr.

NOTE: It is recommended that a cold trap be installed in the roughing line to increase greatly the rate at which water vapor is removed from the insulating materials. It will also prevent the oil from backing up and coming in contact with the insulation around the container.

D. When roughing is completed, switch the fine pumping system into the line. Continue pumping until a stable absolute-pressure of at least 1×10^{-4} Torr has been achieved with the Richards valve closed. While lower pressures are desirable, the pressure reached will depend upon the pumping speed and the ultimate pressure of the system used.

When the evacuation has been completed, close the Richards valve by turning the handle clockwise as far as possible. The handle assembly may then be removed, although leaving it in place will not affect system operation.

NOTE: During the pumpdown procedure, it is desirable to have a vacuum gauge installed in the cooler jacket. A Phillips Gauge Type PHG-09 was found to be sufficiently accurate for the intended purpose and is readily adaptable. Tests have shown that large pressure gradients, five orders of magnitude or more, may exist across the Richards seal-off valve during initial pumpdown and outgassing. Measurements on the downstream side of this valve are not considered to be reliable.

If the pumpdown is performed too slowly or if the desired low pressure cannot be reached, any one of the following are the most likely contributing factors:

1. Contamination of the superinsulation and getter (water vapor, oil, etc.);

2. High impedance of the pumping line;

3. Capacity of pumping system too low; or,

4. Possible vacuum leaks in cooler.

If contamination is indicated, the superinsulation can be "baked out" as will be discussed later, to increase the rate at which contaminants can be removed. Leak checking, when indicated, should be done with a helium mass spectrometer leak detector, if possible.

CAUTION: Dry nitrogen should be used whenever it becomes necessary to raise the jacket pressure to the level of the atmosphere. Ambient air should never be used, as the superinsulation and getter may absorb large amounts of water vapor. When a test device is being installed on the cold finger, the cooler should not be left open any longer than is absolutely necessary.

The pressure in the jacket must never be raised to the level of the atmosphere when the inner container is evacuated (even partially). The coolant container is designed to withstand internal pressure, therefore, reversal of this situation may result in distortion or collapse of the coolant container.

III. BAKE-OUT OF SPACE BETWEEN COOLANT AND OUTER CONTAINERS

If the pumpdown of the vacuum jacket is proceeding too slowly, it may be desirable to raise the temperature of all the components to increase the rate of outgassing. A step-by-step procedure for this bake-out is outlined below:

A. Evacuate Vacuum Jacket

B. Pump

C. Heat Outer Container Carefully Using Heat Gun or Heater Tapes

During tests at the Aerojet laboratory, hot gas for the coolant container was produced by passing dry N_2 gas through an electrical heat exchanger. The current to the heating elements was controlled manually with a Variac and the flow of dry gas controlled by means of a throttling valve and pressure regulator.

IV. PRECOOLING OF COOLANT CONTAINER PRIOR TO FILLING

CAUTION: When using any cryogenic fluid, safety precautions should be exercised to prevent it from coming into contact with either the skin or clothing. The extremely cold fluid will freeze and destroy human tissue causing severe injuries similar to burns. If the coolant container is to be filled with methane or hydrogen, it should be precooled with liquid nitrogen. This is accomplished by transferring several liters of LN_2 into the container and allowing it to boil off. This precooling operation will facilitate subsequent filling. Liquid losses will thereby be reduced during filling and while the filled container is attaining temperature equilibrium. If hydrogen is to be used, all nitrogen must be purged from system with helium gas.

V. FILLING COOLER WITH LIQUID METHANE

A. Be certain that the vacuum jacket is properly evacuated.

B. Make a reliable coupling from the cyrogenic storage container to the 1/4-m. dia. fill line.

C. Begin transferring the liquid. Check to make certain that gas is passing freely from vent line. If no gas issues from vent line, stop filling and check for blockage in the fill and vent lines.

D. When cooler is full, liquid will appear at the exposed end of the vent line. Disconnect the liquid transfer line and close the fill line.

If it is advisable to fill the container only partially. This may be done by monitoring the weight of the cooler during filling and discontinuing the filling operation when the desired coolant weight has been reached. Monitor the gas evolution until temperature equilibrium has been attained. The state of equilibrium is considered as being attained when the gas evolution rate remains constant for 12 hrs. At that time, the cryogenic container may be "topped off" if desired. The gas evolution rate will increase again, but not significantly. Pumping over the liquid coolant for the purpose of solidifying it may now be initiated.

NOTE: Any vessel containing a cryogenic liquid must be allowed to vent at all times. If the fill and vent lines are blocked, the pressure will build up and may ultimately cause the vessel to rupture.

VI. OPERATING THE COOLER CHARGED WITH LIQUID COOLANT

This cooler may be operated with either liquid or solid coolant. When liquid is used, any temperature between the normal boiling point and the freezing point may be attained by controlling the pressure over the liquid. Attempts should not be made to achieve temperatures higher than the normal boiling point by pressurizing the coolant container.

When filled with liquid coolant the vessel should be maintained in the upright position (cold finger down) otherwise the fill and vent lines will be submerged in the coolant which will be forced out as the pressure rises.

The vacuum in the insulation space should never be released when the temperature of the coolant container is being reduced to cryogenic levels. An excessive heat leak into the inner container would result causing rapid pressure rise and discharge, possibly damaging the coolant container and associated equipment. It may also cause condensing of the gas in the insulation space which may damage the insulation.

VII. SOLIDIFICATION OF COOLANT

Solidifying a cryogenic liquid in this cooler entails simply lowering the pressure over the liquid, thereby bringing it to a gentle boil. In doing so, the liquid gives up heat until a new equilibrium between temperature and vapor pressure is reached. Thus, the lower the pressure over the liquid, the lower the temperature will be. For most cryogenic liquids, this process is continuous down to, and past, the freezing point.

Any vacuum pump capable of reasonably high pumping speeds may be used in the solidification process. A water-cooled pump is recommended, as certain gases (notably hydrogen and methane) may cause undue heating in the vacuum pump at high flow rates. Heating will cause considerable loss of pumping efficiency due to the increase in vapor pressure of the oil, coupled with a lowered viscosity. The heating problem is of major significance with hydrogen, moderate with methane, and minor with nitrogen.

The pumping line should be attached to the ¼-in. O.D. Vent port of the cooler. Absolute pressure gauges and/or other accessory equipment may be attached to the fill line for monitoring or controlling the pumpdown rate. The pump and the pumping lines should both have a reasonably high flow rate capacity.

Solidification should be performed at a moderate speed. If the solidification rate is too fast, the liquid has a tendency to "bump" forming a very porous solid. During the pumpdown, it will be noted that the pressure and temperature will drop continuously to a certain value, remaining relatively constant for a time, then continuing to drop. The period of relatively constant pressure and temperature indicates solidification. It is at this point that the rate of pumping should be moderated to achieve as dense a material as possible.

VIII. OPERATION WITH A SOLIDIFIED COOLANT

This cooler may be operated in any position after the coolant has been solidified. The temperature of the cold finger will be directly related to the pressure inside the coolant container, with little dependence on the attitude of the container.

It is desirable, however, to prevent the cooler from undergoing sudden changes in attitude. Since the coolant is now formed into a solid mass, part of which may be in direct contact with the heat transfer rod, a sudden change in attitude might produce internal shock accompanying temperature changes.

When using either methane or nitrogen as a coolant, the pressure (and thus the temperature) may be easily controlled at any desired absolute value by installing a Wallace-Tiernan pressure contactor on the fill line and a solenoid valve in the exhaust (pumping) line. Tests at Aerojet proved that it was easy to maintain temperature within $\pm$ 0.5°K, using this type of control system.

This concludes the operating instructions as applicable to Aerojet Model K6 cryogenic-solid cooler.